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**INFLUENCE OF LIQUID LUBRICANT PROPERTIES
ON THEIR PERFORMANCE**

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ABSTRACT

The influence of lubricant properties on performance is considered in connection with various mechanisms of lubrication. The effects of temperature and pressure on viscosity, which is important in hydrodynamic and elastohydrodynamic lubrication, is presented using a correlation postulated by Roelands. Under elastohydrodynamic conditions it is important to distinguish between the influence of lubricant properties within the inlet region and the Hertz region since each performs different functions. The role of lubricant transport properties such as surface tension is considered in connection with lubricant starvation. Since the lubrication of practical surfaces usually involves boundary as well as hydrodynamic mechanisms, both the chemical and physical properties significantly influence the lubricant's performance.

INTRODUCTION

From a broad point of view lubricants are selected because they perform certain functions. Their performance is intimately related to their intrinsic properties, i.e., chemical composition and molecular structure, but unfortunately for practical reasons these must be expressed in terms of external measured properties. Various lubricant properties and their associated environmental applications are listed in table I.

Since friction wastes power, and wear or scuffing reduces service life, it is seemingly clear that the primary function of a lubricant is to reduce friction and wear. This is generally achieved by the formation of a somewhat nebulous film between the surfaces in motion. This film may be of a "liquid" nature produced by the viscous generation of pressure between the surfaces, or it may be a "solid" developed through physical and chemical adsorption or chemical reactions on the surfaces. In addition to providing a lubricating film, liquid lubricants must perform other functions such as cooling, sealing and transmitting power.

The performance of a lubricant is sometimes difficult to assess since it must function in many ways and under extreme environments. Physical and chemical properties of the rubbing surfaces and their operating conditions such as relative velocity, load and temperature all influence the environment that the lubricant sees. This environment determines the mechanisms of lubrication, usually classified as hydrodynamic, elastohydrodynamic (EHD), mixed and boundary lubrication. Individual efforts in tribological

research, which are frequently concerned with only one type of lubrication, have made significant advances in the understanding of the lubrication mechanisms associated with each particular class of lubrication. In actual liquid lubrication practice there are, however, but few instances where the mechanism of lubrication fits cleanly into one class. In addition, the boundary between each lubrication class is sometimes not very distinct, particularly between those classes where lubricating films are very thin (i.e., EHD, mixed and boundary).

The ability of a lubricant to lubricate depends on how it performs within the environment it encounters (or creates) between the surfaces in motion. This environment may be that found, for example, between the local region of contact between two meshing gear teeth or the region of contact between a ball and a race in a ball bearing. The interactions between the lubricant and its local environment, which may be greatly different than the surroundings, influences its lubricating performance.

The interactions between the lubricant and its environment are shown schematically in figure 1. The performance of a lubricant will depend firstly on the degree of interaction between it and its environment (e.g., how much the operating temperatures change the viscosity). Secondly, the performance will depend on the stability of interaction between the lubricant and its environment (e.g., how this change in viscosity in turn will change the temperature). Higher temperatures can mean lower viscosities and film thicknesses, which through asperity interaction, can create yet higher temperatures. Ultimately, this may lead to surface failure. This is an example of an unstable interaction between the lubricant and its environment. Unstable interactions lead to attrition and failure of the bearing surfaces in the form of wear, scuffing, and fatigue. In the final analysis these are the characteristics by which the performance of a lubricant is judged.

TYPES OF SURFACE ATTRITION OR FAILURE

The types of surface attrition or failure as listed in figure 2 are wear, scuffing and fatigue. Scuffing might be considered a severe form of wear; and, fatigue mechanisms play a role in many types of wear. Their definitions might not, therefore, be clear cut, but these are nevertheless necessary in order to establish a simple correlation between lubricant properties and their failure modes.

Wear

Wear is a somewhat general process, i.e., the interactions between lubricant properties and the environment are reasonably stable. The mechanism of wear may involve corrosion, abrasion, adhesion, and fatigue, or a combination of these.

Corrosive wear occurs when the surfaces actively react with other environmental elements producing either solid and/or gaseous reaction products. Since the reaction product contains material from the surface, attrition of the bearing surfaces proceeds by the continual formation and mechanical removal of the reaction product. Oxygen and moisture generally increase corrosive wear. The chemical properties of the lubricant are therefore important in connection with corrosive wear.

Abrasive wear occurs when hard asperities are pressed into a softer or more brittle surface producing plastic flow and removal of material. Chemical reaction with surface oxides and fatigue mechanisms are usually involved. Both chemical and viscous properties of lubricants are important in connection with abrasive wear.

Adhesive wear occurs when clean or nascent solid surfaces are formed and come into contact. Adhesive junctions are very strong and continual motion frequently causes the material to fracture at weaker subsurface locations in one or both surfaces. Thus, surface attrition occurs by material transfer or removal.

Prevention of adhesion is frequently the most important performance requirement of a lubricant. The viscous properties of a lubricant which determine film thickness can help reduce adhesive wear. Should the viscous properties fail to reduce adhesion, the chemical make-up of the lubricant becomes vitally important. Its ability to form a protective film on the surface may mean the difference between occasional asperity adhesion or scuffing.

Scuffing

Scuffing may be considered as an advanced form of adhesive wear. It is different from ordinary wear in that it is an overall failure of the surface films. It involves gross amounts of adhesion, plastic flow, and material transfer. Scuffing is apparently (or incidentally) associated with temperature through the softening or desorption of surface films [1, 2]. This temperature association is somewhat controversial [1]. A less controversial role of temperature is its influence on the formation of reaction films on the solid surfaces to prevent scuffing. Viscous properties of lubricants influence scuffing by their ability to generate films sufficiently thick to reduce the amount of asperity-asperity contact. On the other hand, viscous heating contributes to the total temperature the local conjunction experiences. Since it is rare that total asperity collision can be eliminated, the performance of a lubricant is intimately related to what the lubricant has to offer on a chemical basis.

Fatigue

If bearing surfaces do not fail by excessive wear or scuffing, they may eventually fail by fatigue. Fatigue is associated with compressive and tensile stress reversals within areas subjected to repeated elastic (or plastic) deformation. The formation of a wear particle initiates from a small surface or subsurface crack. Repeated stress causes the crack to

propagate or grow until a piece of metal is dislodged, leaving a characteristic pit. Fatigue may occur on a microscopic scale or a macroscopic scale. The former relates to local asperities and may also be an active mechanism in wear processes. The latter relates to the total load bearing regions which result in larger fatigue pits of a more serious nature.

Lubricant properties influence fatigue mostly by their ability to form an EHD film of sufficient thickness to eliminate the majority of asperity collisions. Do chemical properties influence fatigue? The answer in most instances is yes. But, the real question is whether its influence is helpful or harmful.

LUBRICATION MECHANISMS

The role that a lubricant takes on in performing its functions varies considerably with the environment it is subjected to. The particular environment encountered determines the mechanisms by which lubricating films are formed. An understanding of these mechanisms is perhaps more important than a knowledge of the available lubricants and their properties.

Hydrodynamic Lubrication

The explanation of the mechanism of hydrodynamic pressure generation might seem somewhat elementary. But, its importance is frequently greater than one realizes and its misunderstanding, especially in the EHD range, may be even more frequent.

The development of hydrodynamic pressure is shown schematically in figure 3. A qualitative and simplified description is as follows. Fluid molecules adjacent to the solid surfaces are assumed to adhere or wet the surface. Thus, the fluid immediately adjacent to the lower surface moves with it at a velocity, u . The velocity of the fluid adjacent to the upper surface is zero since it is stationary. Consider now the velocity distribution between the two surfaces. At a given location the velocity distribution should be linear provided it is not influenced by flow conditions upstream or downstream of that location. This is shown by the dashed lines at the leading and trailing edges of the upper surface. Since the surfaces are not parallel, it would appear that the volume of flow (area inscribed by dashed lines) coming in is greater than that going out. To establish continuity, the fluid flow coming in must be decreased and the fluid flow going out must be increased. This is achieved by the creation of a pressure as shown in figure 3, which resists the flow coming in and helps the flow going out. The amount of pressure needed or generated depends on the resistance of the fluid (viscosity).

The hydrodynamic load capacity is equivalent to the pressure generated which, besides viscosity, is intimately associated with other operating conditions such as relative surface velocity and geometry. The relationship of these variables is described by Reynolds equation which is shown in figure 3. It describes the basic characteristics of hydrodynamic

pressure generation which are: relative motion, r ; converging geometry, $(h-h_0)h^3$ (or a smaller surface separation somewhere downstream), and an interposing fluid with viscosity, μ_0 .

Viscosity is the most important fluid property in Reynolds equation (density is much less significant). In 1916 Martin [3] solved the Reynolds equation in an attempt to explain the absence of wear in gear teeth by the presence of a hydrodynamic oil film. Assuming rigid cylinders and an isoviscous fluid, he obtained the following film thickness equation:

$$\frac{h_0}{R} = 4.896 \frac{u\mu_0}{W} \quad (1)$$

where h_0 is the film thickness on the line of centers, R is the equivalent radius of curvature, W is the load, u is one-half the sum of the surface speeds, and μ_0 is the viscosity at atmospheric pressure. Thus, from the solution of Reynolds equation we see that film thickness is proportional to viscosity under rigid cylinders and isoviscous conditions.

Viscosity versus Structure

Mineral oils are made up of mostly paraffinic, naphthenic, and aromatic hydrocarbon groups as shown in figure 4. These groups are linked together to form a mixture of complex molecules. Synthetic fluids, on the other hand, are much more uniform or orderly in relation to their molecular structure. Their properties are therefore more easily correlated with their molecular structure. While molecular structure may be somewhat complex and varied within a given fluid, certain relations do exist between structure and viscosity. Within a given class of fluids viscosity increases with molecular weight. Straight chain, flexible molecules like paraffin are less sensitive to temperature and pressure than bulky, branched and rigid molecules like naphthenes.

Influence of Temperature on Viscosity

Unlike other material properties which change slightly with temperature, viscosity is somewhat unique in that it can change several orders of magnitude with temperature. This sensitive tie between temperature and viscosity influences performance by way of film thickness ($h_0 \propto \mu_0$) and viscous friction. Figure 5 illustrates Newton's law of viscosity which is a measure of the fluid's resistance to flow; or from a molecular point of view, a measure of the internal friction forces between its moving molecules. The magnitude of this resistance (viscosity) is governed by the forces between and the flexibility of its molecules. Since temperature is a measure of molecular motion, its influence on viscosity is thus very significant. The temperature range seen by the lubricant in a hydrodynamic environment is usually very large, thus making the prediction of the variations of viscosity with temperature important.

Because of the complex molecular structure of lubricating fluids, the approach to viscosity-temperature correlations has been mostly empirical. The standard viscosity-temperature relation is the Walther equation from which the ASTM viscosity-temperature chart (D-341) was formed

$$\log \log (\mu_o + 0.6) = m \log T + C \quad (2)$$

where μ_o = kinematic viscosity
 T = absolute temperature
 m = viscosity-temperature slope
 C = constant, characteristic of fluid

Many lubricating fluids like mineral oils, esters, pure hydrocarbons and silicones give reasonably straight lines on the ASTM chart. Some lubricants, however, do not have constant slopes, particularly at extreme temperature ends.

Possibly a better viscosity-temperature relation is that presented by Roelands [4]. His equation is shown in figure 6. The Roelands correlation is appealing because liquids of the same chemical type give the same viscosity-temperature slope (S) as seen in figure 6. This can therefore be used as a "VI" system for classifying lubricants.

Viscosity-Temperature Improvement

Performance generally would be improved or at least more predictable if the sensitivity between viscosity and temperature were not so great. Lower viscosity-temperature slopes have been achieved with mineral oil by solvent refining, selective blending of paraffinic fluids, and by the addition of relatively small amounts of high molecular weight polymeric additives like polyisobutylene and polymethacrylate esters.

The mechanism of polymer thickening is illustrated in figure 7. It is a function of the solvent ability of the base oil. At low temperatures the base oil is a poor solvent. The internal attractive forces and the intermolecular forces cause the long chain polymers to curl into a tightly rolled up mass of small size. In this configuration it offers relatively little hinderance to the molecular movement of the base oil. At higher temperatures, however, the solvent ability of the base oil improves. The lower intra- and intermolecular forces uncurl and extend the long chain polymers into a configuration which offers much greater hinderance to molecular movement. The greatest viscosity-temperature improvement is achieved when the polymers are completely uncured or extended at the highest environmental temperature and completely curled or near its precipitation point at the lowest environmental temperature.

When polymer thickened oils are subjected to high shear rates, the long chain polymers tend to orientate in the direction of flow causing a reduction in viscosity (temporary viscosity loss). This viscosity loss or non-Newtonian behavior influences performance through the reduction of film thickness and viscous friction. In addition, high shear rates or

cavitation may cause the long chains to mechanically or chemically break down into smaller molecules. This viscosity loss is of a permanent nature which degrades the viscous performance of the lubricant with time. It, therefore, contributes to the instability of the interactions between the lubricant and its environment.

Elastohydrodynamic Lubrication

The interactions between the lubricant and its environment under EHD conditions are somewhat more severe than the hydrodynamic case. The mechanism of pressure generation is essentially the same as in the hydrodynamic case, except for the modifications introduced by the increase in viscosity with pressure and/or the elastic deformation of the bearing surfaces.

EHD Regions and Pressures

A characteristic feature of EHD lubrication is the near Hertzian pressure distribution and elastic deformation. This allows the EHD problem to be divided up into three general regions as shown in figure 8.

The inlet region is a converging section, definitely favorable to the generation of hydrodynamic pressure as described above. The relative importance of the geometry in this region to the generation of pressure is described by the term $(h-h_0)/h^3$ in Reynolds equation. This is shown by the dashed curve in figure 8. It reaches a maximum when the film thickness in the inlet region is approximately $(3/2)h_0$; and it falls to nearly zero in both the upstream and downstream directions when the inlet film thickness is approximately $9h_0$ and h_0 , respectively. Since the geometry term is directly related to pressure generation as described in Reynolds equation (see fig. 8), it must, therefore, follow that the majority of hydrodynamic pressure is generated in the inlet region.

The Hertz region is dominated by the Hertzian pressure. That is, the pressure that would exist if the surfaces were not separated by a lubricant film. The actual pressure differs from this Hertzian pressure only at the leading and trailing edges of the Hertz region. At the leading edge it must blend in with the hydrodynamic pressure of the inlet region. At the trailing edge it must blend in with a secondary hydrodynamic pressure. This secondary pressure is necessary to establish continuity of flow. That is, the very large exit flow produced by the rapidly falling Hertzian pressure must be resisted. This is accomplished by the secondary pressure peak in conjunction with the local exit constriction in the film profile.

The exit region exposes the fluid to a rapidly diverging geometry, definitely favorable to the generation of negative hydrodynamic pressure. Since most fluids degas or cavitate under relatively low tensile stresses, this negative hydrodynamic pressure is almost immediately terminated.

Lubricants Within the EHD Regions

Lubricant properties affect performance under EHD conditions mainly through their influence on the inlet and Hertz regions. The inlet region is of particular significance. The actual pressure reached in this region may be relatively low; but without this hydrodynamic pressure, no film can be established further downstream. This will become evident below in the discussion of starvation. Since film thickness is primarily established in the inlet region, it must therefore follow that performance will be an important function of the interactions between the lubricant and the environment within the inlet region. The most important interaction is that between pressure and viscosity. Fluid molecules see pressures which rapidly rise from ambient to several thousand psi. This increases their flow resistance to nearly one order of magnitude in terms of viscosity.

The solution of the EHD problem has resulted in film thickness equations which are becoming widely used in practice. One such equation for line contact from reference [5] is:

$$\frac{h_m}{R} = 2.65 \left(\frac{\mu_o u}{E' R} \right)^{0.7} (\alpha E')^{0.54} \left(\frac{w}{E' R} \right)^{-0.13} \quad (3)$$

where h_m = minimum film thickness
 R = equivalent radius of curvature
 μ_o = atmospheric viscosity
 α = pressure viscosity coefficient
 u = one half the sum of the surface velocities
 E' = equivalent elastic modulus
 w = load per unit width

In contrast to the hydrodynamic problem where film thickness is directly proportional to viscosity, we see here that under EHD conditions the film thickness is proportional to the fluid properties of viscosity and pressure-viscosity raised to the powers of 0.7 and 0.54, respectively. The above equation indicates that film thickness is not very sensitive to load. Recent experimental work [6, 7], however, has shown that film thickness is much more sensitive to load under highly loaded conditions. It is possible that this may be explained by some fluid rheological behavior which has not been included in the theory.

There may be other factors that can cause the film thickness to be lower than that predicted from equation (3). The fluid molecules see a great deal of shear within the inlet region due to the converging geometry. The flow spectrum is illustrated in figure 9 for pure rolling. It can be shown that the maximum shear stress occurs at a location within the inlet region where the film thickness is approximately $2h_o$. This is close to the heart of the film generating region. Thus, any non-Newtonian fluid behavior will surely influence film thickness. Finally, the fluid molecules can be affected by the generation of heat within the inlet region [8].

This can occur under very high speed conditions where viscous heating becomes significant in the portions of the inlet region where backflow and swirling occur (fig. 9).

The interactions between the lubricant and its environment in the inlet region influences its ability to pump the lubricant film up. The interactions in the Hertz region, however, influences the way it rides this lubricant film. Once an initial film has been achieved in the inlet region, it is very difficult for it to escape while passing through the Hertz region. This is the result of two major conditions. The first is the very high pressures and viscosities which can give the fluid the consistency of butter or cheese. The second is the very large film extent compared to its thickness, i.e., the width of the film may be two or three orders of magnitude greater than its thickness. These two conditions provide an enormous resistance to fluid flow, particularly in the central portion of the Hertz region where the pressures are high and pressure gradients are low. The very large flow resistance that exists within the Hertz region is illustrated in figure 10 which is a photomicrograph of an oil film which has been trapped between a steel ball loaded statically against a glass substrate. The oil film was initially formed under rolling conditions. When the rolling motion was terminated, part of the oil film remained. The thickness of the oil film is illustrated by the interference fringes which were obtained using green light. The surfaces are in contact (or very nearly so) only along the edge of the Hertz region. The surfaces within the central portion of the Hertz region are separated by an unsymmetrical film of a few micro inches thick which remained for a considerable period of time. This is an indication that the fluid under these conditions has a solid-like consistency.

The enormous resistance to flow in the Hertz region is the major cause of friction or traction. In addition to the effects of pressure on viscosity, temperature effects are significant in traction at all but the lowest slip speeds. At high pressures and temperatures the correlation between traction and the Newtonian stress-strain relation breaks down [9]. Fluid flow characteristics under these severe stress conditions are largely unknown.

Influence of Pressure on Viscosity

The interaction between pressure and fluid viscosity is important to the success of an EHD lubricating film. Measurements of viscosity under high pressures are difficult to make, and the correlation of the results in a simple form for the many different types of fluids has not been entirely satisfactory.

The variation of viscosity with pressure is usually described by an exponential relation proposed by Barus (see ref. [10]) who used it to correlate his results on marine glue. The relation is

$$\mu = \mu_0 e^{\alpha p} \quad (4)$$

where μ = viscosity at pressure p
 μ_0 = viscosity at atmospheric pressure
 α = pressure-viscosity coefficient

An example of this type of correlation is shown in figure 11 where pressure is plotted against log viscosity. A straight line with constant slope for all temperatures indicates a good correlation with the exponential relation. Naphthenic oils tend to follow this relation better than paraffinic oils, but both show decreasing slopes with temperature. Synthetic fluids as shown in figure 12 do not follow the exponential relation much better.

Although the pressure-viscosity coefficient (α) may vary with both temperature and pressure for a given fluid, it nevertheless is somewhat successfully incorporated into the EHD film thickness equations [11]. Its success is partially due to the fact that pressure-viscosity coefficients do not vary more than threefold among most fluids. Also, choosing a pressure-viscosity coefficient somewhat judiciously for the controlling pressure and temperature range involved is important. The inlet region, for example, usually sees pressures ranging from ambient to not more than twenty or thirty thousand psi and temperatures close to that of their surfaces. Several ways of calculating a pressure-viscosity coefficient from pressure-viscosity data are shown schematically in figure 13. These involve the use of slopes and tangents to the pressure viscosity curve at particular pressures or pressure ranges as shown in figure 13(a). Figure 13(b) illustrates an integral method due to Blok, see reference [4].

A better correlation between pressure and viscosity is that of Roelands [4]. His correlation is particularly useful for comparing pressure-viscosity characteristics of fluids. The Roelands equation is presented in figure 14. Also, the pressure-viscosity coefficient (Z) as a function of temperature for several fluids is plotted in that same figure. All fluids except the synthetic hydrocarbon show very little variation with temperature. This particular synthetic hydrocarbon has very high traction capabilities; therefore, unique pressure-viscosity characteristics are not surprising.

The advantage of the Roelands pressure-viscosity coefficient (Z) over the exponential pressure-viscosity coefficient (α) is that it remains sensibly constant with temperature. It cannot, however, be used directly in the film thickness equation. It is possible, however, to calculate the α value used in the film thickness equation from the Z value and atmospheric viscosity (μ_0) at a particular temperature [4]. This is particularly useful since it provides a good estimation of the α value as a function of temperature from pressure-viscosity data of only one isotherm.

Lubricant Transport Properties and Starvation

It frequently happens (or is required) that the inlet region is not completely flooded with lubricant. The lubricant distribution around the contact region might look like that shown in figure 15. As the distance S , measured from the edge of the Hertz region to the inlet lubricant boundary approaches zero, a lesser amount of inlet pressure is allowed to build up. The primary result is a reduction in film thickness. This is shown in figure 16 where film thickness and starvation S are plotted in dimensionless parameters. In addition, the film shape and, therefore, the pressure distribution approach the Hertzian condition for dry contact. This is shown by the interference fringes in the photomicrographs of figure 17. In figure 17(a) the boundary distance is sufficiently removed from the edge of the Hertz region to obtain a flooded condition. Figure 17(b-f) show a diminishing film thickness as the inlet lubricant boundary progressively approaches the edge of the Hertz region. As starvation progresses the EHD shape approaches the Hertz shape. The EHD pressure must therefore approach the Hertz pressure. In the limit; i.e., when the film thickness is zero, the Hertz shape and pressure are attained.

The influence of speed (u), viscosity (μ_0) and pressure-viscosity coefficient (α) on film thickness under starvation conditions is shown in figure 18. When the inlet lubricant boundary is far removed from the Hertz region (large S/R), the film thickness increases with speed, viscosity and pressure-viscosity coefficient along the curve marked "flooded". On log-log paper this would be a straight line with a slope of about 0.7. As starvation becomes more severe, the film thickness becomes less sensitive to the surface speed as well as the hydrodynamic fluid properties of viscosity and pressure-viscosity as we usually think of them. Practically, it frequently happens that S decreases with speed. This can lead to a degenerating situation where film thickness decreases, rather than increases, with speed. Film thickness then becomes more sensitive to the variables affecting the lubricant transport properties and how they influence the amount of flooding in the inlet region.

The theoretical results of figure 18 are in line with recent performance data on a gyroscope bearing which was known to be operating under starved conditions [12]. Here it was found that increasing speed had much less influence on film thickness than EHD theory would predict. Viscosity appeared to be the most important single variable. It is also an important variable in connection with lubricant transport. In addition to viscosity, surface tension is another important lubricant transport property.

The role of surface tension can be illustrated by considering two lubricated surfaces (e.g., a ball loaded on a flat) in static contact as shown in figure 19. Figure 19(a) shows the shape of the lubricant-air interface for a hypothetical case of a zero-surface tension fluid. A real fluid with a finite surface tension will cause a pressure differential across an interface wherever the surface is curved. The pressure differential Δp is given by

$$\Delta p = \frac{\gamma_{LV}}{r_b} \quad (5)$$

where γ_{LV} is the surface tension, and r_L is the effective radius of curvature at the lubricant boundary. To establish equilibrium, the interface will tend to approach an infinite radius of curvature, and thus form a fluid meniscus around surfaces of close proximity (see fig. 19(b)). In this way surface tension should aid the filling of an EHD inlet region. Under dynamic conditions the shape of the fluid meniscus will be affected by the distribution of lubricant on the bearing surfaces and by the pressure and flow fields around the contact region (see fig. 15). This includes the lubricant being discharged in the exit region where cavitation occurs as shown by the photomicrograph in figure 20. The depletion of lubricant within a bearing track due to cavitation is shown schematically in figure 21. The effectiveness of surface tension to recapture and maintain lubricant in the inlet region will, according to the above equation, depend on the curvature of the lubricant boundary and the surface tension of the lubricant.

The effect of temperature on surface tension for various lubricants is shown in figure 22. Surface tension decreases slightly with temperature, but can vary considerably from lubricant to lubricant. The low surface tension of the fluorinated polyether is significant. It has been found to be far more prone to starvation than other fluids [13]. This may be the result of its low surface tension and also its high density.

The effect of temperature on surface tension is almost insignificant compared to its effect on viscosity. Therefore, the influence of surface tension on lubricant transport will become more significant as the environmental temperature increases.

Boundary Lubrication

The surface energy of atomically clean solids can be very high [14]. In this state they become chemically non-selective and react with almost anything they come in contact with. It is well documented that adhesion between two clean solids can be very great, and that adhesion plays an important role in surface failure mechanisms. The traditional explanation of the function of a boundary lubricating film is that it provides an adsorbed molecular layer of low surface energy on the solid surface. The physical properties of this adsorbed film and lubricating mechanisms are not entirely clear, but its formation and performance on solid surfaces has been the subject of many papers. The influence of lubricant properties on performance in the boundary region is a function of the chemical makeup of the lubricant.

Constitution of Boundary Environment

A typical solid surface in contact with a lubricant is shown in figure 23. Most solid surfaces in an air atmosphere contain films of oxides and adsorbed water. The physical and chemical nature of these films influence the interactions between the lubricant and the solid surface.

Lubricating fluids such as mineral oils contain small amounts of materials such as fatty acids which have attached polar (head) and hydrocarbon (tail) groups. Some hydrocarbon groups may contain chemically active atoms like sulfur, oxygen, and nitrogen. Also, mineral oils will contain small amounts of dissolved air and water. Additive oils may contain a variety of other elements which can participate in chemical reactions. These active components of lubricating fluids and their interactions with their environment determine lubrication performance.

Lubrication with Boundary Films

When lubricant is interposed between two surfaces as shown in figure 23, the polar components such as fatty acids, alcohols, amines and esters absorb on the solid surfaces. The association of the polar (head) groups with the solid surface may be of a physical or chemical nature. Physically adsorbed polar molecules are weakly held by van der Waals' molecular forces of attraction. This is an equilibrium process. That is, molecules continually attach and detach themselves from the surface. The hydrocarbon groups (tails) are laterally held together by cohesive forces which give the adsorbed film strength and rigidity. The physical properties of this film are thus markedly different than in the bulk fluid. In the adsorbed state they can have solid-like properties. One of the worst enemies of a physically adsorbed film is high temperature, which can cause it to desorb entirely. Adsorption may frequently be physical at low temperature and chemical at higher temperature. A chemically adsorbed film is stronger and utilizes bonds of a chemical nature.

An ideal mechanism for the lubrication of adsorbed films is shown in figure 24 where the interface between two opposing films provides a low friction shear plane between the sliding surfaces. The actual lubricating film, however, may not be made up entirely of the surface active material, but may be a mixture of surfactant (e.g., fatty acid) and hydrocarbon of the base fluid. Evidence of the influence the base fluid solvent has on the performance of adsorbed boundary films [15] is shown in figure 25. The lubricating ability is greatest when the polar component and hydrocarbon carrier have equal chain lengths.

High temperatures from the heat generated by plastic deformation and fluid shear cause the boundary film to desorb, disorientate, or melt. This can lead to local adhesion between solid surfaces as shown in figure 26. Lubricating ability is therefore frequently measured by the temperature (transition temperature) at which the coefficient of friction rises. In general, physically and chemically adsorbed boundary films are effective primarily at low and moderate temperatures.

Effective lubrication at higher temperature can be achieved if various components of the fluid chemically react with the solid surfaces to form reaction products with suitable lubricating properties. These reaction products are generally formed from the components of the fluid such as organics or organometallics containing chlorine, sulfur or phosphorous. These components may occur naturally within the fluid such as the sulfur commonly found in mineral oils, or they may be of the additive type such

as EP (extreme pressure) additives. These additives are supplied in specific concentrations and compositions to satisfy particular performance requirements. Since chemical reactions are involved, the formation of an inorganic EP film is sensitive to temperature, time, reactivity of the solid surfaces, chemical composition and concentration of the additive. Suitable environmental conditions that promote the formation of an EP film can be created by the rubbing process itself. That is, local asperity contact can produce the required temperature and solid surface exposure for reaction to take place. This is shown schematically in figure 27 where a reaction film has formed at a location where the surface was abraded by asperity contact.

Lubrication by an EP film is a sacrificial process; i.e., the EP film formed is continually wiped away by the rubbing process. The reaction must be rapid enough to form a protective film between each successive removal, but yet slow enough to prevent excessive corrosion.

While boundary films may be solid-like, what about the interface between the boundary film and the bulk liquid? There is evidence of a somewhat ordered, greasy, or highly viscous layer that can exist between the boundary film and the bulk liquid [16]. In addition, chemical reactions between lubricants and bare metal surfaces apparently can form a viscous colloidal fluid [17]. The effect of pressure on these viscous materials is unknown, but could be important, and the possibility of a local EHD contribution to boundary-type lubrication exists.

Mixed Lubrication

The thickness of EHD lubricating films found in practice are usually on the order of the individual asperity roughnesses. Thus, the load in many practical applications may be supported partially by a fluid film and partially by local areas of asperity contact. The lubrication of these areas of contact are important since they can influence friction and initiate failure. It has been found, for example, that the friction which arises from the interactions between asperities (i.e., boundary friction) between two steel disks is on the order of one-half to two-thirds the total friction when scuffing is likely to occur [18]. Thus, the boundary lubricating properties of the fluid must be a significant factor affecting performance under mixed EHD lubrication conditions. The influence of fluid film forming properties of the fluid such as viscosity and pressure viscosity, however, are by no means insignificant. These properties are perhaps more important under conditions where asperity interactions occur than when they do not occur since film thickness determines the degree of asperity interaction [19].

Surface roughness can influence performance by its effect on the inlet region and also the Hertz region. In the inlet region it can influence the generation of hydrodynamic pressure. It has been shown [19] that surface roughness orientated in the longitudinal direction only can inhibit pressure generation and therefore film thickness. On the other hand, transverse roughness can enhance pressure generation thus leading to slightly thicker films than with ideally smooth surfaces. Real surface roughnesses, however, are generally not entirely one-dimensional. Thus the total effect of roughness on film thickness will usually be small.

The influence of surface roughness in the Hertz region is much more significant since asperity contact will occur here first. The degree of asperity contact depends on the magnitude of the surface roughness compared to the film thickness. This is usually expressed as a ratio (h_0/σ) where h_0 is the film thickness and σ is the combined roughness of both surfaces. Since film thickness is determined in the inlet region where the effect of surface roughness is small the EHD theory assuming smooth surfaces can be used to calculate the film thickness provided the load carried by the asperities is not a substantial portion of the total load.

Under mixed EHD conditions the film thickness as well as the asperity heights are normally small compared to the overall elastic deformation of the surfaces which is similar to the Hertz deformation. The total pressure distribution will therefore be similar to the Hertz pressure distribution. Furthermore, under mixed EHD conditions, the total pressure distribution is the sum of the fluid film pressure and the asperity pressure. It has been shown [19] that the asperity pressure is determined directly by the parameter (h_0/σ). This explains why the parameter (h_0/σ) appears to be significant in connection with surface attrition through wear and fatigue [20].

Scuffing, which is a much more severe form of surface attrition, is no doubt also related to the parameter (h_0/σ). Recent work has indicated that the rate of production of heat by friction per unit Hertz area of contact is an important criterion in scuffing failure [21]. The rate of heat generated is a function of the friction due to the interaction between asperities and also the viscous friction due to fluid shear. Since the former is related to boundary type lubrication and the latter to EHD lubrication, it is not surprising that both the chemical and physical properties of fluids have important influence on performance in practical applications.

INDIRECT LUBRICATING PROPERTIES

There are a number of lubricant properties that can influence its ability to lubricate in an indirect way. Some of these properties are: heat capacity, thermal conductivity, freezing point, vapor pressure, gas solubility, foaming, corrosion, thermal stability and oxidation stability. Although these have not been considered above, it is important to note that in certain applications these indirect lubricating properties may be primarily responsible for the lubricant's performance. An example of this is the high temperature application of jet engine lubricants where the particular lubricant chosen may have been primarily based on its oxidation and thermal stability.

CONCLUSIONS

Lubricant performance is intimately related to its chemical composition and molecular structure. For practical reasons these must be expressed in terms of externally measured properties like viscosity, heat capacity, gas solubility, oxidation stability, etc. The performance of a lubricant

is complicated by the variety and severity of the environment it sees; thus, the measured properties are only significant so long as they can be correlated with the behavior in its local environment.

Since friction wastes power, and wear or scuffing reduces service life, it is seemingly clear that the primary function of a lubricant is to reduce, or at least control, the same. This is generally achieved by the formation of a somewhat nebulous phase or film between surfaces in motion. This film, which carries the load and bears the shear between the surfaces, may be of a liquid nature produced by the generation of hydrodynamic pressure, or it may be of a solid nature developed through adsorption or chemical reactions on the surfaces (or both). The lubricant performs its functions by way of the mechanisms that produce these lubricating films. The controlling mechanism, which depends on the environment in which the lubricant functions, may be classified as hydrodynamic, elastohydrodynamic (EHD), mixed and boundary lubrication. It is important to consider these mechanisms if one is to understand the influence of lubricant properties on performance.

In hydrodynamic lubrication the lubricant interposed between converging surfaces in relative motion can generate pressure capable of supporting substantial loads. The hydrodynamic pressure generated is a direct function of lubricant viscosity. Since viscosity is very sensitive to temperature, a means of correlating viscosity-temperature characteristics of lubricants is important. The usual method is the Walther equation which forms the basis of the ASTM viscosity charts. Perhaps a better correlation is that of Roelands which behaves consistently for a wide range of fluid structures. Improvement of viscosity-temperature characteristics can be achieved by adding long chain polymers. The viscosity of these lubricants, however, can be quite sensitive to shear rates.

Elastohydrodynamic lubrication is characterized by high local pressures and elastic deformations which are distinctly similar to the Hertzian condition for dry contact. These results in three major regions which perform different functions with the lubricant. The "inlet region" pumps the film up; the "Hertz region" rides it; and the "exit region" discharges it. In addition to temperature, pressure has an important influence on viscosity in the elastohydrodynamic environment. The level of viscosity in the inlet region influences film thickness, and in the Hertz region it influences traction. Pressure-viscosity correlation using an exponential relation is frequently used since it is convenient for theoretical calculations. Its accuracy with pressure-viscosity data varies with pressure and temperature; and, from lubricant to lubricant. A Roelands correlation has better accuracy but is less convenient in theoretical calculations.

If the inlet region is not properly flooded with lubricant, the lubricating film will become starved, i.e., the film becomes thinner. Bearing design and lubricant supply systems are important considerations in connection with the starvation problem; but lubricant transport properties such as surface tension, density, and viscosity can also be very influential. Surface tension can vary appreciably among synthetic fluids and it has been observed that a fluorinated polyether which has low surface tension and high density is somewhat prone to starvation.

In boundary lubrication the chemical composition of lubricants is as important to the formation of solid-like boundary films as viscosity is to the formation of thicker liquid films. Surface active materials such as fatty acids can form physically and chemically adsorbed films of molecular dimensions on solid surfaces. In addition, chemical reactions between lubricant components and the surrounding materials can form a mixture of inorganic and organic films of considerable thickness compared to an adsorbed film. The formation and composition of boundary films are not clearly understood, and even less is known about their mechanical properties and the mechanisms by which they lubricate. But, their ability to separate opposing high surface energy solids and thus prevent adhesion or welding is probably the greatest supporting role a lubricant can offer to lubrication.

Mixed lubrication involves both the mechanisms of elastohydrodynamic (and/or hydrodynamic) and boundary lubrication. It is an important area of lubrication since the thicknesses of fluid films frequently found in practice are on the order of the individual asperity roughnesses. Thus, the load may be supported partially by an elastohydrodynamic film and partially by a boundary film. The thickness of the elastohydrodynamic film relative to the surface roughness directly influences the asperity pressure. Since surface attrition in the form of wear, fatigue and scuffing is intimately associated with local asperity contact both the physical and chemical properties of the fluid have significant influence on performance.

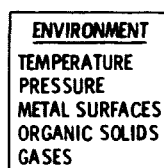
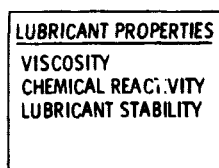
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**TABLE I. - LUBRICANT PROPERTIES AND
AREAS OF IMPORTANCE**

Property	Environmental application
Foaming	Gears and circulation
Bulk modulus	High pressure
Vapor pressure	High temperature and/or vacuum
Miscibility	Compatibility - additives and seals
Gas solubility	Cavitation, boundary lubrication
Heat capacity	Heat transfer
Thermal conductivity	Heat transfer
Freezing point	Starting
Viscosity and V. I.	Hydrodynamic lubrication
Pressure viscosity	Elastohydrodynamic lubrication
Work of adhesion	Boundary lubrication
Thermal stability	Inert atmospheres
Oxidation stability	Ordinary atmospheres
Corrosion	Metals



WEAR (ABRASIVE, CORROSIVE, ADHESIVE)

SCUFFING

FATIGUE

PERFORMANCE A FUNCTION OF:

1. Degree of interaction between the lubricant and the environment.
2. Stability of the interaction.

Figure 2. - Types of surface attrition or failure.

CS-64825

Figure 1. - Interaction between lubricant and environment.

CS-64833

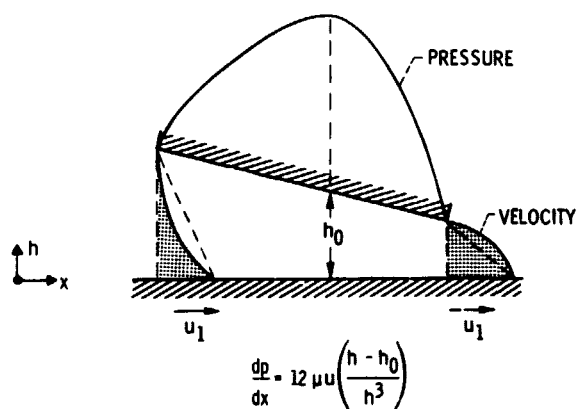


Figure 3. - Formation of hydrodynamic pressure, a function of viscosity (μ) as well as velocity (u) and geometry $(h - h_0)/h^3$.

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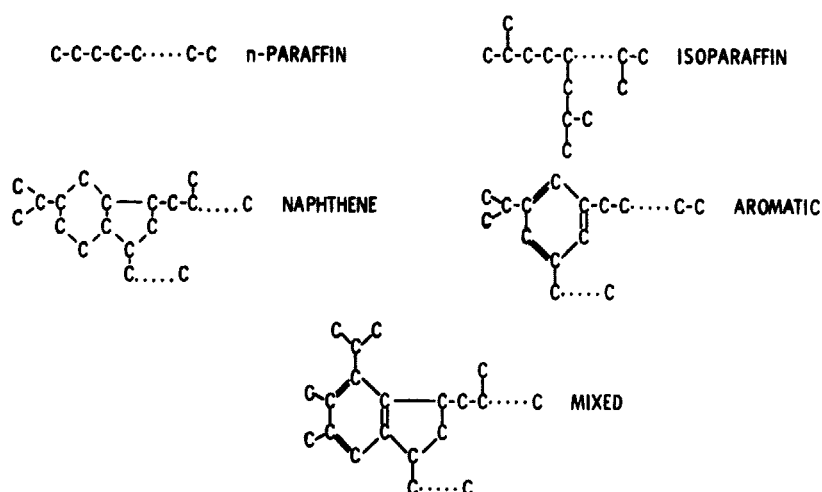
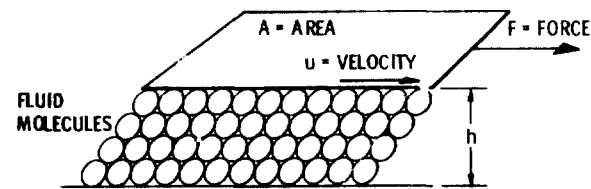


Figure 4. - Molecular structure of typical mineral oils.

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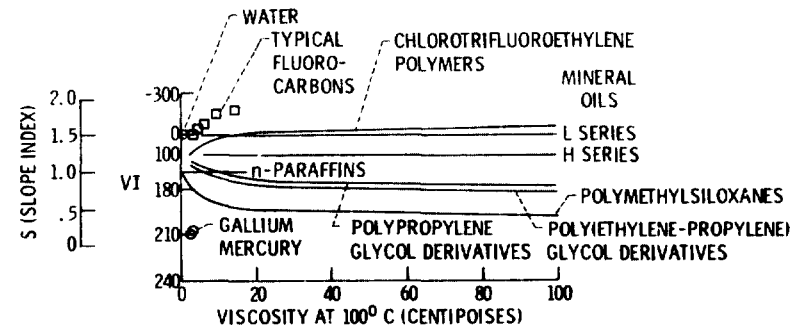
$$F = \mu \frac{AU}{h}$$

OR

$$\mu = \frac{F/A}{u/h} = \frac{\text{SHEAR STRESS}}{\text{SHEAR RATE}}$$

Figure 5. - Newton's law of viscosity.

CS-64839



$$\text{LOG}(1.200 + \text{LOG } \mu) = \text{LOG } G - S \text{ LOG}(1 + t_c/135)$$

WHERE

μ = VISCOSITY IN CENTIPOISES

t_c = TEMPERATURE IN °C

G = PARAMETER INDICATING VISCOSITY LEVEL

S = MEASURE OF VARIATION OF VISCOSITY WITH TEMPERATURE (V. I. INDICATOR)

CS-64819

Figure 6. - Viscosity-temperature correlation of Roelands.

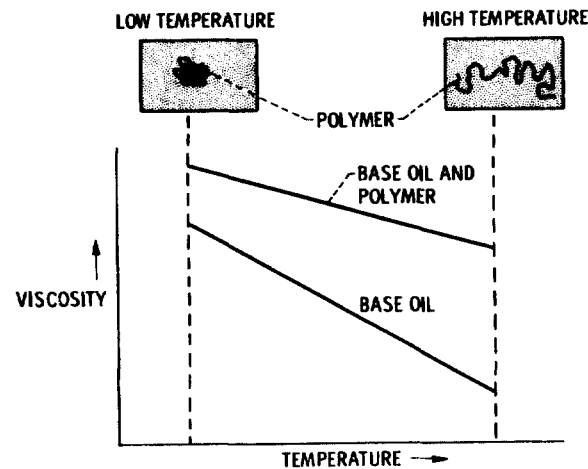
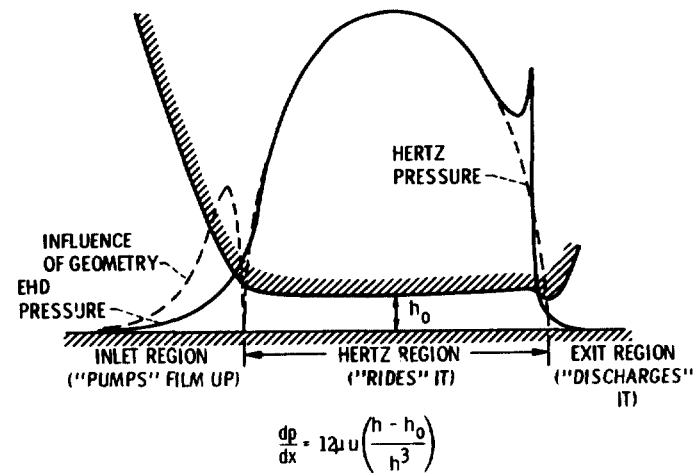


Figure 7. - Mechanism of polymer thickened oil.

CS-64826



$$\frac{dp}{dx} = 12\mu u \left(\frac{h - h_0}{h^3} \right)$$

Figure 8. - EHD pressure generation.

CS-64827

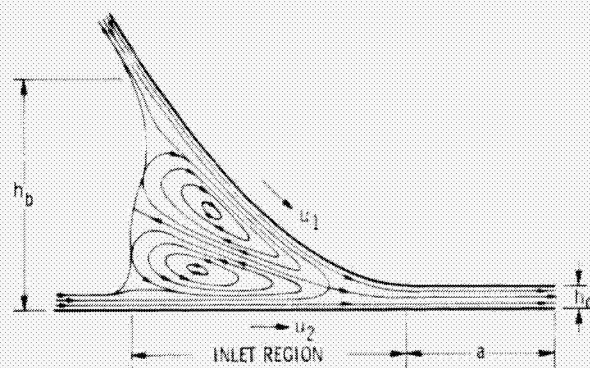


Figure 9. - Flow spectrum in inlet region.

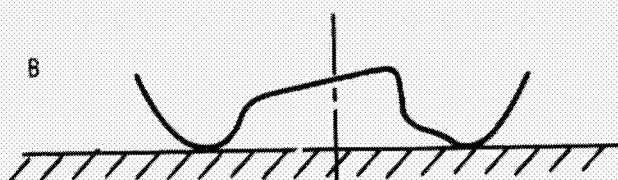
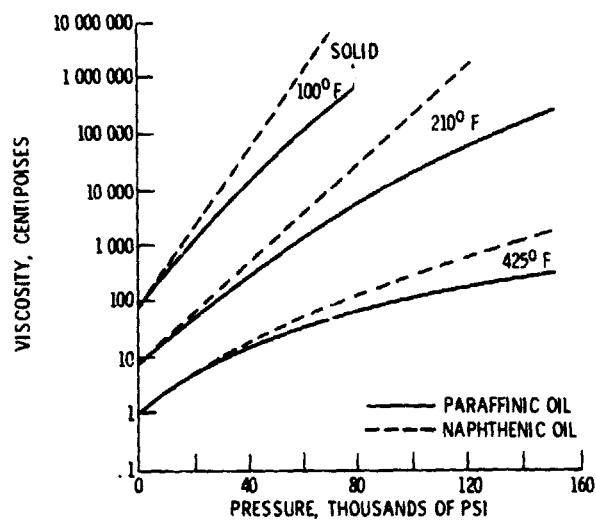


Figure 10. - (a) Photomicrograph of an oil entrapment; (b) schematic representation of film thickness.

E-7151

CS-64815



CS-64820

Figure 11. - Effect of pressure on viscosity for paraffinic and naphthenic oils at different temperatures.

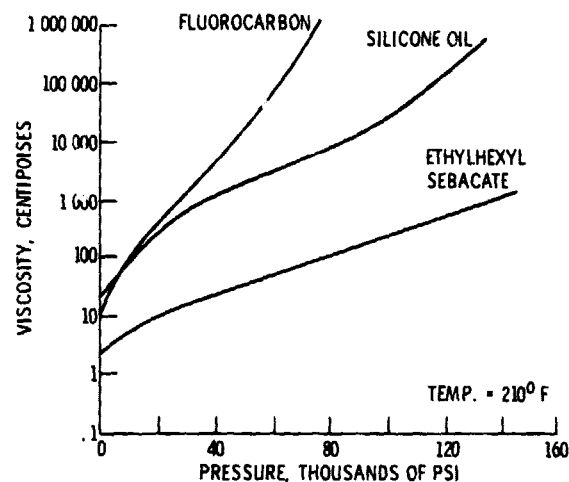
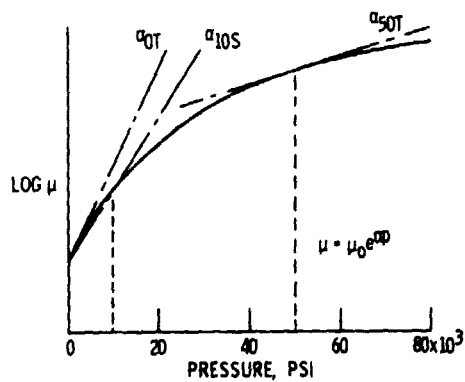
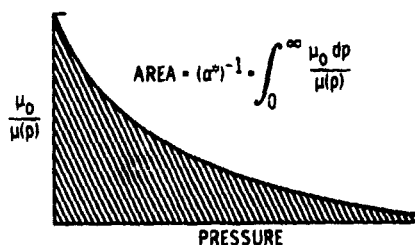


Figure 12. - Effect of pressure on viscosity for several synthetic oils.

CS-64829



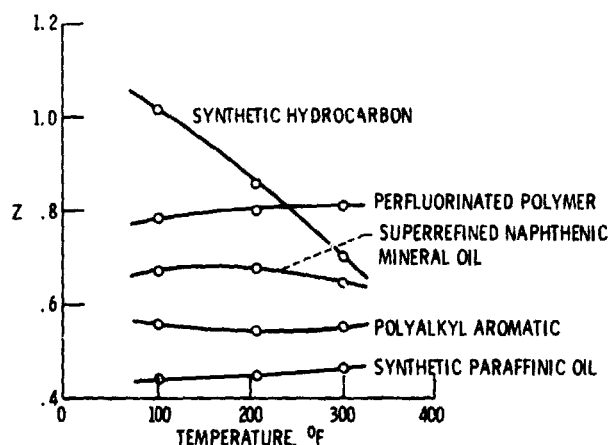
(a) TANGENT AND SLOPE METHODS.



(b) INTEGRAL METHOD (BLOK).

Figure 13. - Schematic representations of pressure-viscosity characteristics (a).

CS-64822



$$\text{LOG}(\text{LOG } \mu + 1.200) = Z \text{ LOG} \left(1 + \frac{P}{28.4} \right) + 4$$

WHERE

μ = VISCOSITY IN CENTIPOISE

P = PRESSURE IN THOUSANDS OF PSI

Figure 14. - Roelands' pressure-viscosity characteristic as a function of temperature.

CS-64821

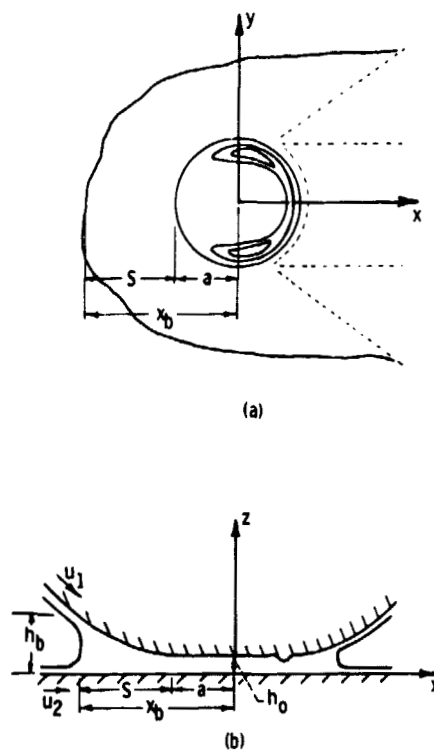


Figure 15. - Contact geometry showing required measurements for starvation; (a) plan view, (b) cross-sectional view along centerline.

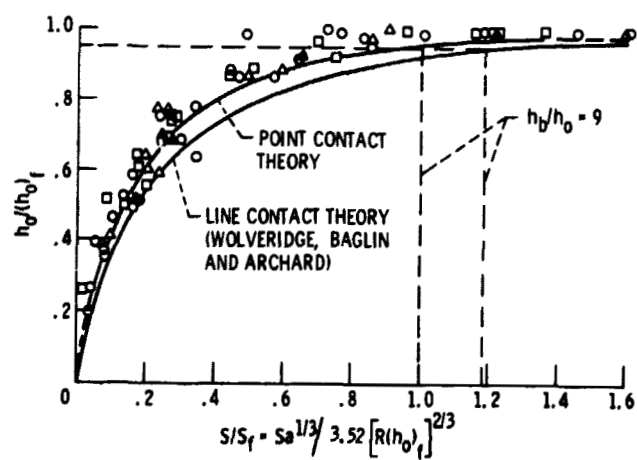


Figure 16. - Effect of degree of inlet flooding on film thickness for line and point geometry.

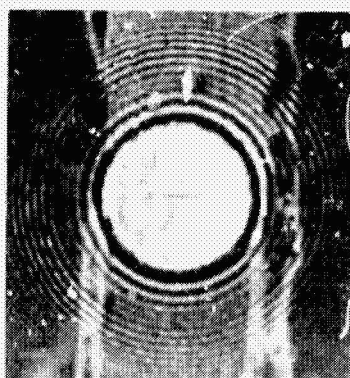
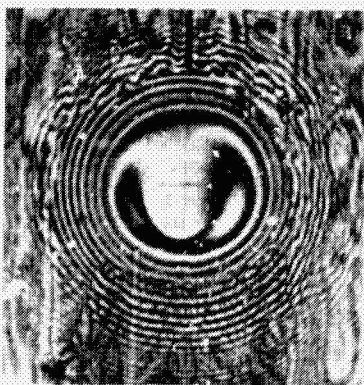
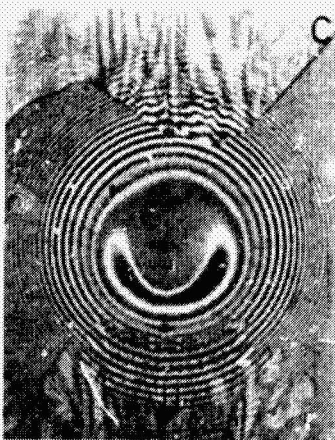
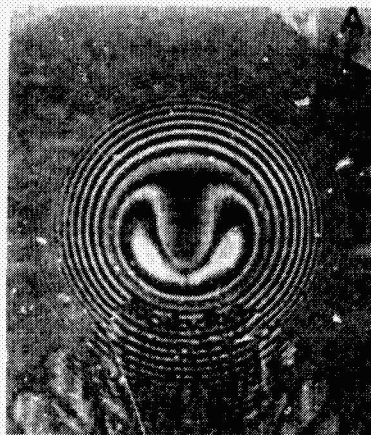


Figure 17. - Photomicrographs showing progressive starvation.

CS-64813

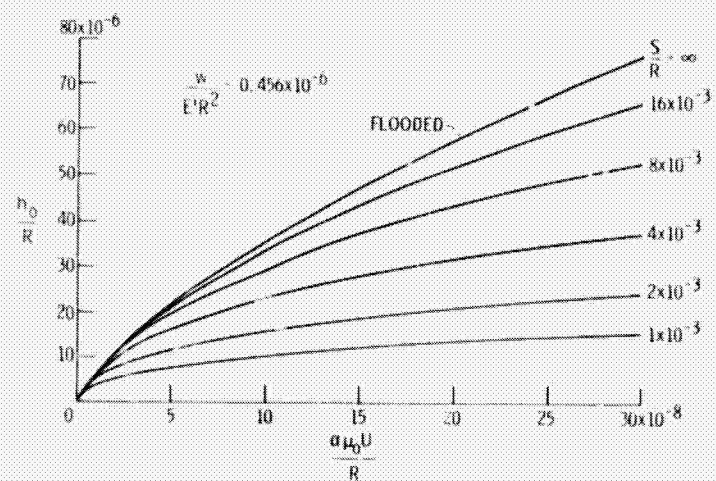


Figure 18. - Effect of speed-viscosity parameter on film thickness for constant degrees of starvation.

CS-64823

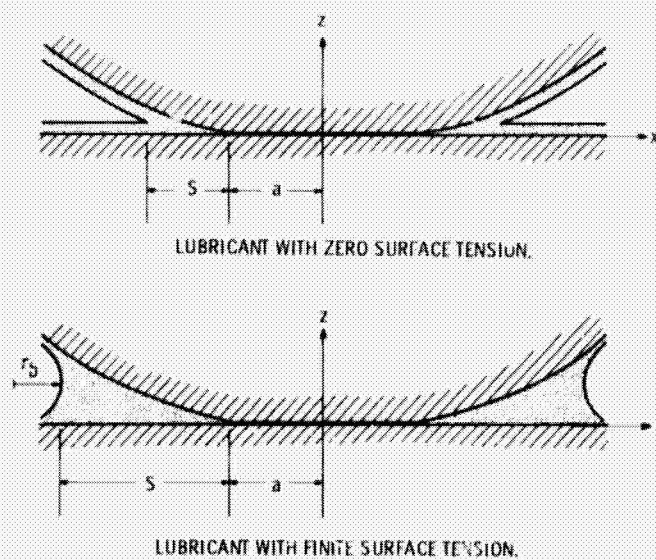


Figure 19. - Contact geometry under static conditions.

CS-64823



Figure 20. - Cavitation in exit region of point contact geometry in pure rolling.

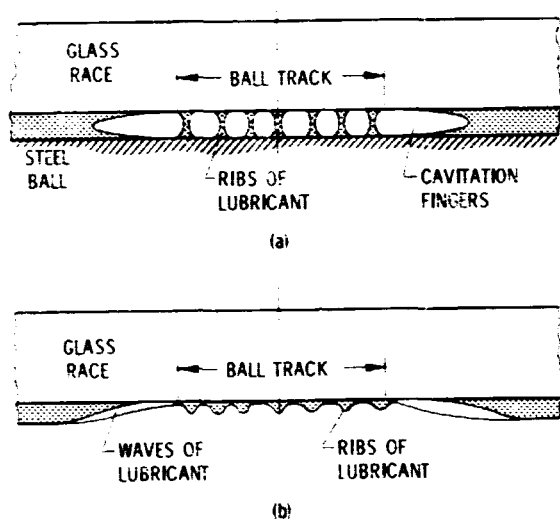


Figure 21. - Schematic representation of the cavitated region: (a) at a constant radius from the wall center, and (b) showing the distribution of lubricant after the ball has passed.

CS-44824

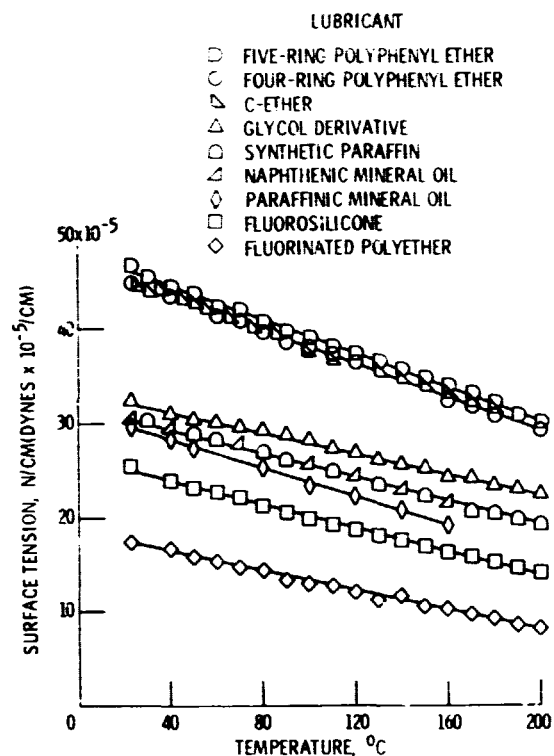


Figure 22. - Surface tension as function of temperature for lubricants.

CS-44817

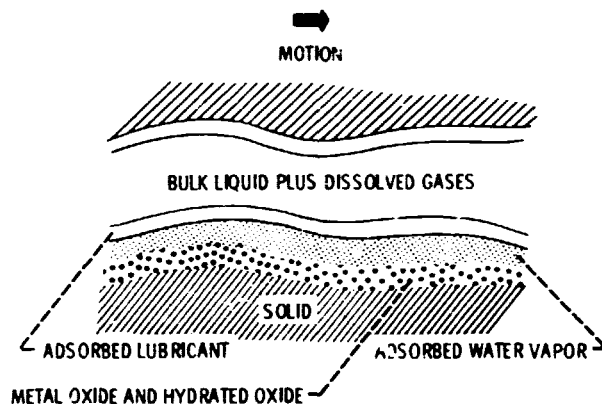


Figure 23. - Undisturbed surfaces.

CS-44837

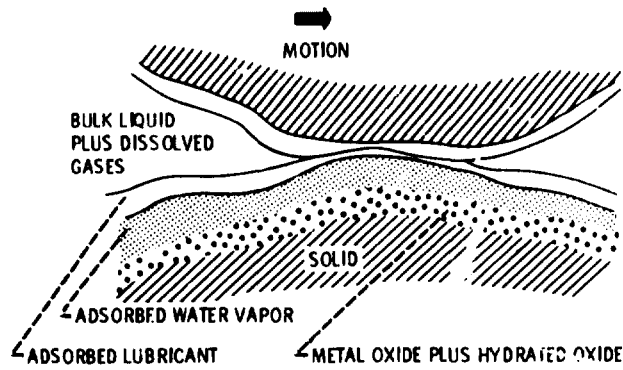


Figure 24. - Lubrication by adsorbed lubricant. CS-64838

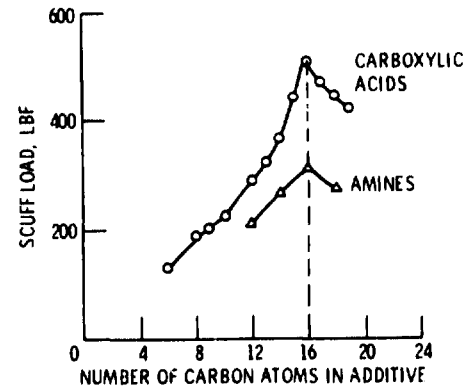


Figure 25. - Effect of chain length on lubricating ability for 1/3 mol % additive concentration in cetane ($C_{16}H_{34}$). CS-64826

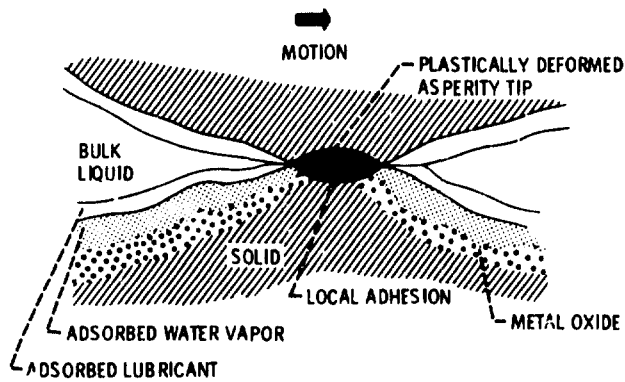


Figure 26. - Failure of surface films. CS-64836

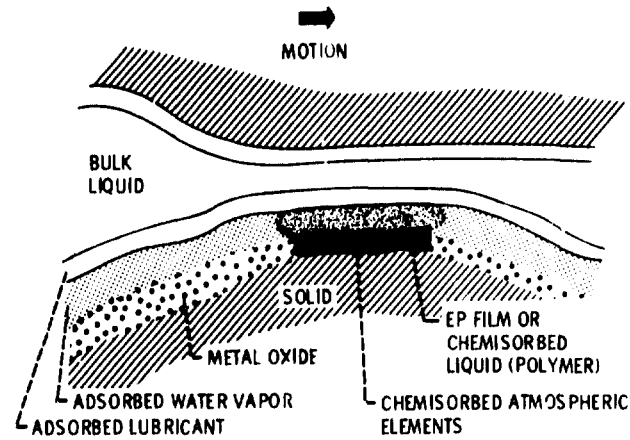


Figure 27. - Abraded surface after contact. CS-64835